Notice No.4

Rules and Regulations for the Classification of Ships, July 2019

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Please note that corrigenda amends to paragraphs, Tables and Figures are not shown in their entirety.

Issue date: November 2019

Amendments to	Effective date	IACS/IMO implementation (if applicable)
Part 1, Chapter 2, Section 2	Corrigendum	NA
Part 1, Chapter 3, Sections 9, 15, 21 & 24	Corrigenda	NA
Part 3, Chapter 1, Section 9	Corrigenda	NA
Part 3, Chapter 3, Section 5	Corrigenda	NA
Part 3, Chapter 9, Section 9	Corrigenda	NA
Part 3, Chapter 13, Section 2	Corrigendum	NA
Part 3, Chapter 14, Section 1	Corrigenda	NA
Part 4, Chapter 2, Sections 3 & 5	Corrigenda	NA
Part 5, Chapter 1, Section 1	Corrigenda	NA
Part 5, Chapter 2, Sections 7, 10, 12 & 15	Corrigenda	NA
Part 5, Chapter 3, Section 7	Corrigenda	NA
Part 5, Chapter 6, Section 3	Corrigenda	NA
Part 5, Chapter 10, Sections 2 – 11 & 14	Corrigenda	NA
Part 5, Chapter 11, Sections 2 – 7, 10 & 11	Corrigenda	NA
Part 5, Chapter 12, Sections 2, 4, 5, 7 & 8	Corrigenda	NA
Part 5, Chapter 13, Sections 1 & 2	Corrigenda	NA
Part 5, Chapter 14, Sections 4 & 10	Corrigenda	NA
Part 5, Chapter 15, Sections 3 & 4	Corrigenda	NA
Part 5, Chapter 16, Section 7	Corrigenda	NA
Part 6, Chapter 3, Sections 1 & 4	Corrigenda	NA
Part 7, Chapter 10, Sections 6 & 9	Corrigenda	NA
Part 7, Chapter 15, Sections 2, 3 & 5	Corrigenda	NA

Lloyd's Register

Part 1, Chapter 2
Classification Regulations

■ *Section 2*
Character of classification and class notations

2.1 Definitions

Table 2.2.2 Special features notations

Special features notation	Description	*See also*
For Liquefaction and Storage of (Methane, etc.) in Independent Gas Tanks (Type B, etc.), Maximum Vapour Pressure ( ) ~~bar~~ MPa, Minimum Temperature Minus ( ) °C	Assigned where ships of Category 1B or 2 which have process plants installed solely for the purposes of the physical liquefaction of impure feedstock gases at low temperature and the storage of the purified liquefied gases (where the chemical treatment of the impurities is an incidental process)	*Pt 7, Ch 2, 2.2 Additional notations*

Part 1, Chapter 3
Periodical Survey Regulations

■ *Section 9*
Ships for liquefied gases

9.6 Intermediate Surveys

9.6.2 In addition to the requirements for Annual Survey and the requirements of *Pt 1, Ch 3, 3.3 Intermediate Surveys 3.3.1* to *Pt 1, Ch 3, 3.3 Intermediate Surveys ~~3.3.8~~ 3.3.9*, the following are to be dealt with as applicable:
(d) Where required by the manufacturer's maintenance instructions, cargo tank and inter-barrier space pressure and vacuum relief valve settings are to be checked and adjusted as required. Cargo tank pressure relief valve harbour settings are also to be checked, if applicable. Cargo tank pressure relief valves are to lift at a pressure not more than the percentage given below, above the maximum vapour pressure for which the tanks have been approved.

- For ~~0 to 1,5 bar (0 to 1,5 kgf/cm^2)~~, 0 to 0,15 MPa, 10 per cent.
- For ~~1,5 to 3,0 bar (1,5 to 3,0 kgf/cm^2)~~, 0,15 to 0,3 MPa, 6 per cent.
- For pressures exceeding ~~3,0 bar (3,0 kgf/cm^2)~~, 0,3 MPa, 3 per cent.
- Valves may be removed from the tanks for the purpose of checking

■ *Section 15*
Boilers

15.1 Frequency of surveys

15.1.1 All boilers, economizers, steam receivers, steam heated steam generators, thermal oil and hot water units intended for essential services, together with boilers used exclusively for non-essential services having a working pressure exceeding ~~3,5 bar~~ 0,35 MPa and a heating surface exceeding 4,5 m^2 are to be surveyed internally. There is to be a minimum of two internal examinations during each five-year Special Survey cycle. The interval between any two such examinations is not to exceed 36 months. A general external examination is to be carried out at the time of the Annual Survey.

■ *Section 21*
Refrigerated cargo installations

21.3 Subsequent Special Survey

21.3.4 Shell-and-tube evaporators (secondary refrigerant coolers) in which the secondary refrigerant is in the shell are to have the shell hydraulically tested (with the end covers removed) to 1,5 times the design pressure, but in no case less than ~~2,9 bar g~~ 0,29 MPa. After refitting the end covers, the primary refrigerant side is to be pneumatically tested as stated in *Pt 1, Ch 3, 21.3 Subsequent Special Survey 21.3.3*, and an examination made as far as practicable for gas leakage in the shell with the secondary refrigerant connection removed.

■ *Section 24*
Fuel installations using gases or other low-flashpoint fuels

24.8 Complete Surveys – General requirements

24.8.10 Relief valves are to be surveyed as follows:
(b) Valves are to lift at a pressure not more than the percentage given below, above the maximum vapour pressure for which the tanks have been approved:

- For ~~0 to 1,5 bar,~~ 0 to 0,15 MPa, 10 per cent.
- For ~~1,5 to 3,0 bar,~~ 0,15 to 0,3 MPa, 6 per cent.
- For pressures exceeding ~~3,0 bar,~~ 0,3 MPa, 3 per cent.

Part 3, Chapter 1
General

■ *Section 9*
Procedures for testing tanks and tight boundaries

9.7 Definitions and details of tests

9.7.3 **Hose test** is a test used to verify the tightness of joints with a jet of water. The jet of water is to be directed perpendicular to the joint. It is to be carried out with the pressure in the hose nozzle maintained at not less than ~~2,0 bar~~ 0,2 MPa during the test. The hose nozzle is to have a minimum inside diameter of 12 mm and is to be situated no further than 1,5 m from the joint. Where a hose test is not practical because of possible damage to machinery, electrical equipment insulation or outfitting items, it may be replaced by a careful visual examination of welded connections, supported by an ultrasonic or penetration leak test, or an equivalent, *see* SOLAS *Regulation 11 - Initial testing of watertight bulkheads, etc.*~~.~~

9.7.4 **Tank air test** is to be used to verify the tightness of a compartment by means of an air pressure differential and leak indicator solution. An efficient indicating solution (e.g. soapy water) is to be applied to the weld or penetration being tested and is to be examined whilst an air pressure differential of not less than ~~0,15 bar~~ 0,015 MPa is applied by pumping air into the compartment. Arrangements are to be made to ensure that any increase in air pressure does not exceed ~~0,30 bar~~ 0,03 MPa. A U-tube with a height sufficient to hold a head of water corresponding to the required test pressure is to be used for verification and to avoid overpressure. The cross-sectional area of the U-tube is not to be less than that of the pipe supplying air to the tank. Alternatively, two calibrated pressure gauges may be considered acceptable. All boundary welds, including pipe connections in the compartment are to be examined twice. The first is to be examined immediately upon applying the leak indication solution; the second approximately five minutes afterwards.

9.7.5 **Compressed air fillet weld test**. This test consists of compressed air being injected into one end of a fillet welded joint and the pressure verified at the other end of the joint by a pressure gauge. Pressure gauges are to be arranged so that an air pressure of at least ~~0,15 bar~~ 0,015 MPa above atmospheric pressure can be verified at each end of all passages within the portion being tested. A leak indicator solution is to be applied and the weld line examined for leaks. A compressed air test may be carried out for partial penetration welds where the root face is greater than 6 mm.

9.7.6 **Vacuum box test** is a test used to verify the tightness of joints by means of a localised air pressure differential and leak indicator solution. The test is to be conducted with the use of a box with air connections, gauges and an inspection window that is to be placed over the joint being tested with a leak indicator solution applied. The air within the box is to be removed by an ejector to create a vacuum i.e. a pressure differential of ~~0,20 to 0,26 bar~~ 0,02 to 0,026 MPa inside the box.

Part 3, Chapter 3
Structural Design

■ *Section 5*
Design loading

5.4 Design pressure for partially filled tanks

5.4.4 The critical fill range, *F*~~-~~$_{\text{crit}}$, is to be determined using the following formula:

$$\cancel{F_{\text{crit}} = \left(\frac{100}{H_{\text{t}}}\right)\left[\left(\frac{L_{\text{s}}}{2\pi}\right) ln\left(\frac{1+\eta}{1-\eta}\right) + b\sqrt{\frac{n}{n+1}}\right]\%} \qquad F_{\text{crit}} = \left(\frac{100}{H_{\text{t}}}\right)\left[\left(\frac{L_{\text{s}}}{2\text{π}}\right) ln\left(\frac{1+\text{η}}{1-\text{η}}\right) + b\sqrt{\frac{n}{n+1}}\right]\%$$

where

ln = natural logarithm to base e

$\eta = \cancel{\frac{4\pi L_s}{[(S_{\mathrm{nr}}-5)2g]}}$ $\frac{4\pi L_s}{[(S_{\mathrm{nr}}-5)\ \ ^2 g]}$ for fill level at S_{nr} – 5 seconds upper bound roll critical fill level

or $\eta = \cancel{\frac{4\pi L_s}{[(S_{\mathrm{nr}}+5)2g]}}$ $\frac{4\pi L_s}{[(S_{\mathrm{nr}}+5)\ \ ^2 g]}$ for fill level at S_{nr} + 5 seconds lower bound roll critical fill level

or $\eta = \cancel{\frac{4\pi L_s}{[(S_{\mathrm{np}}-3)2g]}}$ $\frac{4\pi L_s}{[(S_{\mathrm{np}}-3)\ \ ^2 g]}$ for fill level at S_{np} – 3 seconds upper bound pitch critical fill level

or $\eta = \cancel{\frac{4\pi L_s}{[(S_{\mathrm{np}})2g]}}$ $\frac{4\pi L_s}{[(S_{\mathrm{np}})\ \ ^2 g]}$ for fill level at S_{np} seconds

Part 3, Chapter 9
Special Features

■ *Section 9*
Strengthening for machinery on deck

9.2 Loading

Table 9.9.1 Ship motions

<table>
<tr><td>Motion</td><td colspan="2"></td><td>Acceleration</td></tr>
<tr><td colspan="4">Symbols</td></tr>
<tr><td colspan="2">C_w = a wave head, in metres
= 7,71 x $10^{-2} L e^{-0,0044L}$ but L is not to be taken greater than 227
$k_m = \cancel{1+\frac{k_r(0,5-x_m)}{(C_b+0,2)}}$ $1+\frac{k_r(0,5-x_m)^2}{(C_b+0,2)}$</td><td colspan="2">$K = \cancel{13\frac{B}{GM}}$ $13\ \frac{GM}{B}$ but is not to be taken less than 1</td></tr>
</table>

Part 3, Chapter 13
Ship Control Systems

■ *Section 2*
Rudders

2.15 Cone couplings with special arrangements for mounting and dismounting the couplings

2.15.6 In case of hydraulic pressure connections the required push-up force P_e, for the cone may be determined by the following formula:

$\cancel{P_e = P_{\mathrm{req}} d_m \Pi l\left(\frac{c}{2} + 0{,}02\right) N}$ $P_e = P_{\mathrm{req}} d_m \Pi l\left(\frac{\theta_t}{2} + 0{,}02\right) N$

Part 3, Chapter 14
Cargo Securing Arrangements

■ *Section 1*
General

1.7 Symbols and definitions

1.7.1 The following definitions are applicable to this Chapter, except where otherwise stated:

x_c = longitudinal centre of gravity of a container forward of the ~~AP~~ aft end of *L*, in metres

x_{om} = longitudinal centre of motion forward of the ~~AP~~ aft end of *L*. To be taken at the LCF of the ship

LCF = longitudinal centre of flotation. To be taken ~~as~~ at 0,48*L* from the aft end of *L*

Part 4, Chapter 2
Ferries, Roll On-Roll Off Ships and Passenger Ships

■ *Section 3*
Deck structure

3.4 Deck supporting primary structure

3.4.2 For ferries and passenger ships where direct calculations have been carried out in accordance with the SDA procedure relevant to the ship type (including multi-decked ships), the section modulus of primary members supporting four or more point loads or a uniformly distributed load is not to be taken as less than:
~~$Z = 0{,}673\ S\ Ps\ k\ l_{e2}$ (cm^3)~~ $Z = 0{,}673 S P_s k l_e^2$ (cm^3)

■ *Section 5*
Shell envelope framing

5.2 Strengthening for wave impact loads

5.2.3 The effective section properties of secondary stiffeners are to be taken as:

(a) Plastic section modulus of secondary stiffeners, Z_p is to be taken as:

Z_p = ~~$(2{,}8 \times 10^{-4}\ s_{cm}\ t_p^{\ 2}) - (10^{-3}\ b_f\ b_{fc}\ t_f \sin\theta_e) + 5 \times$~~
~~=~~ ~~$10^{-4}\ (h_w^{\ 2}\ t_w + 2 b_f\ t_f\ h_w) \cos\theta_e$ cm^3~~

$(2{,}8 \times 10^{-4}\ s_{cm}\ t_p^{\ 2}) - (10^{-3}\ b_f\ b_{fc}\ t_f \sin\theta_e) + (5 \times 10^{-4}\ (h_w^{\ 2}\ t_w + 2 b_f\ t_f\ h_w) \cos\theta_e)$ cm^3

where

b_{fc} = $0{,}5(b_f - t_w)$ for L profiles
= 0 for flat bar and T profiles
= $\frac{c}{2}$ for bulb profiles, *see Figure 4.7.1Dimensions of longitudinals* in *Pt 3, Ch 4 Longitudinal Strength* ~~for bulb profiles~~ for c

Part 5, Chapter 1
General Requirements for the Design and Construction of Machinery

■ *Section 1*
General

1.1 Machinery equipment to be constructed under survey

1.1.1 Unless an alternative approach for product assurance has been approved by LR (*see Pt 5, Ch 1, 1.3 Alternative approach for product assurance*), all items of machinery and equipment in ships built under Special Survey are to be surveyed at the manufacturer's works. The Surveyor is to be satisfied that the workmanship is of a suitable standard and that the components are suitable for the intended purpose and duty. Examples of such items are:

- Boilers supplying steam for propulsion or for services essential for the safety or the operation of the ship at sea, including superheaters, economisers, desuperheaters, steam heated steam generators and steam receivers. All other boilers having working pressures exceeding ~~3,4 bar (3,5 kgf/cm^2),~~0,34 MPa, and having heating surfaces greater than 4,65 m^2.
- Valves and other components intended for installation in pressure piping systems having working pressures exceeding ~~7 bar~~ 0,7 MPa.

Part 5, Chapter 2
Reciprocating Internal Combustion Engines

■ *Section 7*
Control and monitoring of main, auxiliary and emergency engines

7.3 Auxiliary engine governors

7.3.1 Auxiliary engines intended for driving electric generators are to be fitted with governors which, with fixed setting, are to control the speed within 10 per cent momentary variation and 5 per cent permanent variation when the full load is suddenly taken off or, when after having run on no-load for at least 15 minutes, the load is suddenly applied as follows:

(a) For engines with BMEP less than ~~8 bar,~~ 0,8 MPa, full load, or

(b) For engines with BMEP greater than ~~8 bar,~~ 0,8 MPa, $\frac{800}{\text{BMEP}}$ per cent, but not less than one third, of full load, the full load being attained in not more than two additional equal stages as rapidly as possible.

■ *Section 10*
Safety arrangements

10.1 Relief valves

10.1.2 Crankcases are to be provided with lightweight spring-loaded valves or other quick-acting and self-closing devices to relieve the crankcases of pressure in the event of an internal explosion and to prevent any inrush of air thereafter. The valves are to be designed and constructed to open quickly and be fully open at a pressure not greater than ~~0,2 bar~~ 0,02 MPa.

■ *Section 12*
Turbochargers

12.4 Matching with engine

12.4.2 Category C turbochargers used on propulsion engines are to be tested to ensure an adequate operating margin without surge occurring during the engine works trials as specified below. These tests may be waived if successfully tested earlier on an identical configuration of engine and turbocharger (including same nozzle rings).

(b) For crosshead engines the surge margin is to be demonstrated by at least one of the following methods:

(ii) Sudden fuel shut-off to at least one cylinder is to not result in continuous surging and the turbocharger(s) is/are to stabilise at the new load within 20 seconds. For applications with more than one turbocharger the fuel is to be shut-off to the cylinders immediately upstream of each turbocharger. This test is to be performed at two different engine loads:

- The engine load corresponding to a charge air pressure of about ~~0,6 bar~~ 0,06 MPa (but without auxiliary blowers running).

■ *Section 15*
Engines supplied with low pressure gas

15.9 Gas ignition in the cylinder

15.9.1 The requirements are as described in:

- *Rules and Regulations for the Construction and Classification of Ships for the Carriage of Liquefied Gases in Bulk, July 2019*, *Ch 16, 16.7 Special requirements for gas-fired internal combustion engines*~~, as applicable~~.

Part 5, Chapter 3
Steam Turbines

■ *Section 7*
Tests and equipment

7.3 Hydraulic tests

7.3.2 The cylinders of all turbines are to be tested to 1,5 times the working pressure in the casing, or to ~~2,0 bar (2,0 kgf/cm^2),~~ 0,2 MPa, whichever is the greater.

7.3.4 Condensers are to be tested in the steam space to ~~1,0 bar (1,0 kgf/cm²)~~ 0,1 MPa. The water space is to be tested to the maximum pressure which the pump can develop at ship's full draught with the discharge valve closed plus ~~0,7 bar (0,7 kgf/cm²),~~ 0,07 MPa, with a minimum test pressure of ~~2,0 bar (2,0 kgf/cm²)~~ 0,2 MPa. Where the operating conditions are not known, the test pressure is to be not less than ~~3,4 bar (3,5 kgf/cm²),~~0,34 MPa, *see Pt 5, Ch 14 Machinery Piping Systems*.

Part 5, Chapter 6
Main Propulsion Shafting

■ *Section 3*
Design

3.9 Bronze or gunmetal liners on shafts

3.9.6 Each continuous liner or length of liner is to be tested by hydraulic pressure to ~~2,0 bar~~ 0,2 MPa after rough machining.

3.12 Sternbushes

3.12.1 The length of the bearing in the sternbush next to and supporting the propeller is to be as follows:

(b) For water lubricated bearings lined with two or more circumferentially spaced sectors of an approved plastics material, in which it can be shown that the sectors operate on hydrodynamic principles, the length of the bearing is to be such that the nominal bearing pressure will not exceed ~~5,5 bar~~ 0,55 MPa. The length of the bearing is to be not less than twice its diameter.

(d) For bearings which are white-metal lined, oil lubricated and provided with an approved type of oil sealing gland, the length of the bearing is to be approximately twice the diameter required for the screwshaft and is to be such that the nominal bearing pressure will not exceed ~~8,0 bar~~ 0,8 MPa. The length of the bearing is to be not less than 1,5 times its diameter.

Part 5, Chapter 10
Steam Raising Plant and Associated Pressure Vessels

■ *Section 2*
Cylindrical shells and drums subject to internal pressure

2.1 Minimum thickness

2.1.1 Minimum thickness, *t*, of a cylindrical shell is to be determined by the following formula:

~~$t = \frac{pR_i}{10\sigma J - 0{,}5\mathrm{p}} + 0{,}75\ \mathrm{mm}$~~ $t = \frac{pR_i}{\sigma J - 0{,}5\mathrm{p}} + 0{,}75\ \mathrm{mm}$

2.4 Unreinforced openings

2.4.2 The maximum diameter, *d*, of any unreinforced isolated openings is to be determined by the following formula:

~~$d = 8{,}08[D_o t(1 - K)]^{1/3}$ *in mm*~~ $d = 8{,}08[D_o t(1 - K)]^{1/3}$ in mm

The value of *K* to be used is calculated from the following formula:

~~$K = \frac{pD_o}{18{,}2\sigma\ t}$ *but is not to be taken as greater than 0,99*~~ $K = \frac{pD_o}{1{,}8\sigma\ t}$ but is not to be taken as greater than 0,99

■ *Section 3*
Spherical shells subject to internal pressure

3.1 Minimum thickness

3.1.1 The minimum thickness of a spherical shell is to be determined by the following formula:

~~$t = \frac{pR_i}{20\sigma J - 0{,}5\mathrm{p}} + 0{,}75\ \mathrm{mm}$~~ $t = \frac{pR_i}{2\sigma J - 0{,}5\mathrm{p}} + 0{,}75\ \mathrm{mm}$

■ *Section 4*
Dished ends subject to internal pressure

4.1 Minimum thickness

4.1.1 The thickness, *t*, of semi-ellipsoidal and hemispherical unstayed ends, and the knuckle section of torispherical ends, dished from plate, having pressure on the concave side and satisfying the conditions listed below, is to be determined by the following formula:

~~$t = \frac{pD_oK}{20\sigma J} + 0{,}75 \text{ mm}$~~ $t = \frac{pD_oK}{2\sigma J} + 0{,}75 \text{ mm}$

4.1.4 In addition to the formula in *Pt 5, Ch 10, 4.1 Minimum thickness 4.1.1* the thickness, *t*, of a torispherical head, made from more than one plate, in the crown section is to be not less than that determined by the following formula:

~~$t = \frac{pR_i}{20\sigma J - 0{,}5\text{p}} + 0{,}75 \text{ mm}$~~ $t = \frac{pR_i}{2\sigma J - 0{,}5\text{p}} + 0{,}75 \text{ mm}$

■ *Section 5*
Conical ends subject to internal pressure

5.2 Minimum thickness

5.2.1 The minimum thickness, *t*, of cylinder, knuckle and conical section at the junction and within the distance, *L*, from the junction is to be determined by the following formula:

~~$t = \frac{pD_oK}{20\sigma J} + 0{,}75 \text{ mm}$~~ $t = \frac{pD_oK}{2\sigma J} + 0{,}75 \text{ mm}$

5.2.3 The minimum thickness, *t*, of those parts of conical sections not less than a distance, *L*, from the junction with a cylinder or other conical section is to be determined by the following formula:

~~$t = \frac{pD_c}{(20\sigma J - p)} \frac{1}{\cos \alpha} + 0{,}75 \text{ mm}$~~ $t = \frac{pD_c}{(2\sigma J - p)} \frac{1}{\cos \alpha} + 0{,}75 \text{ mm}$

■ *Section 6*
Standpipes and branches

6.1 Minimum thickness

6.1.2 For boilers having a working pressure exceeding ~~50 bar~~ 5 Mpa and safety valves of full lift or full bore type, the thickness of the branch pipe carrying the superheater or drum safety valves is to be not less than:

■ *Section 7*
Boiler tubes subject to internal pressure

7.1 Minimum thickness

7.1.1 The minimum wall thickness of straight tubes subject to internal pressure is to be determined by the following formula:

~~$t = \frac{pD_o}{20\sigma + p}$ mm~~ $t = \frac{pD_o}{2\sigma + p}$ mm

■ *Section 8*
Headers

8.2 Rectangular section headers

8.2.1 The thickness of the flat walls of rectangular section headers is to be determined at the centre of the sides, at all the lines of holes and at the corners. The minimum required is to be the greatest thickness determined by the following formula:

~~$t = \frac{pn}{20\sigma J} + \sqrt{\frac{0{,}4Y_P}{\sigma J_1}} + 0{,}75\, mm$~~ $t = \frac{pn}{2\sigma J} + \sqrt{\frac{4Y_P}{\sigma J_1}} + 0{,}75 \text{ mm}$

8.3 Toroidal furnace headers

8.3.1 The minimum thickness of a toroidal header forming the lower end of a waterwall furnace, and supporting the weight of the boiler and water, is to be determined by the following formula:

where

~~$A = \frac{pr}{30J\sigma}$ mm~~ $A = \frac{pr}{3J\sigma}$ mm

~~$M = \frac{W\mathrm{r}}{3} - \frac{pd2\mathrm{r}}{40}$Nmm~~ $M = \frac{W\mathrm{r}}{3} - \frac{pd^2\mathrm{r}}{4}$Nmm

8.4 Header ends

8.4.3 Ends attached by welding are to be designed as follows:

- Flat ends: the minimum thickness of flat end plates is to be determined by the following formula:

~~$t = d_\mathrm{i}\sqrt{\frac{pC}{\sigma}} + 0{,}75\ \mathrm{mm}$~~ $t = d_\mathrm{i}\sqrt{\frac{10pC}{\sigma}} + 0{,}75\ \mathrm{mm}$

■ *Section 9*
Flat surfaces and flat tube plates

9.1 Stayed flat surfaces

9.1.6 The thickness, *t*, of those portions of flat plates supported by stays and around tube nests is to be determined by the following formula:

~~$t = Cd\sqrt{\frac{P}{\sigma}} + 0{,}75\ \mathrm{mm}$~~ $t = Cd\sqrt{\frac{10P}{\sigma}} + 0{,}75\ \mathrm{mm}$

9.2 Combustion chamber tube plates under compression

9.2.1 The thickness of combustion chamber tube plates under compression due to the pressure on the top plate, based on a compressive stress not exceeding 96 N/mm^2 is to be determined by the following formula:

~~$t = \frac{pWs}{1930(s-d)}$ mm~~ $t = \frac{pWs}{193(s-d)}$ mm

9.3 Girders for combustion chamber top plates

9.3.2 The thickness of steel plate girders supporting the tops of combustion chambers is to be determined by the following formula:

~~$t = \frac{0{,}32pl2s}{d2R_{20}}$mm~~ $t = \frac{3{,}2pl^2s}{d^2R_{20}}$ mm

9.4 Flat plate margins

9.4.1 The width, *b*, of a flat plate which may be regarded as being supported by the shell, furnaces or flues to which the flat plate is attached is not to exceed that determined by the following formula:

~~$b = C(t - 0{,}75)\sqrt{\frac{\sigma}{p}}$ mm~~ $b = C(t - 0{,}75)\sqrt{\frac{\sigma}{10p}}$ mm

9.4.5 In no case is the diameter *D*, in mm, of the circle forming the boundary of the margin supported by the uptake of a vertical boiler to be greater than determined by the following formula:

~~$D = \sqrt{\frac{345A}{P} + d2}$~~ $D = \sqrt{\frac{34{,}5A}{P} + d^2}$

■ *Section 10*
Flat plates and ends of vertical boilers

10.1 Tube plates of vertical boilers

10.1.1 Where vertical boilers have a nest or nests of horizontal tubes, so that there is direct tension on the tube plates due to the vertical load on the boiler ends or to their acting as horizontal ties across the shell, the thickness of the tube plates in way of the outer rows of tubes is to be determined by the following formula:

~~$t = \frac{2pD}{JR_{20}}$+ 0,75 mm~~ $t = \frac{20pD}{JR_{20}}$+ 0,75 mm

10.2 Horizontal shelves of tube plates forming part of the shell

10.2.1 For vertical boilers of the type referred to in *Pt 5, Ch 10, 10.1 Tube plates of vertical boilers*, in order to withstand vertical load due to pressure on the boiler ends, the horizontal shelves of the tube plates are to be supported by gussets in accordance with the following formula:

~~$C = \frac{10AD_ip}{t}$~~ $C = \frac{100AD_ip}{t}$

10.3 Dished and flanged ends for vertical boilers

10.3.1 The minimum thickness, *t*, of dished and flanged ends for vertical boilers which are subject to pressure on the concave side and are supported by central uptakes is to be determined by the following formula:

~~$t = \frac{10pR_i}{13\sigma} + 0,75\ \text{mm}$~~ $t = \frac{100pR_i}{13\sigma} + 0,75\ \text{mm}$

■ Section 11

Furnaces subject to external pressure

11.2 Corrugated furnaces

11.2.1 The minimum thickness, *t*, of corrugated furnaces is to be determined by the following formula:

~~$t = \frac{pD_o}{C} + 0,75\text{mm}$~~ $t = \frac{10pD_o}{C} + 0,75\ \text{mm}$

11.3 Plain furnaces, flue sections and combustion chamber bottoms

11.3.1 The minimum thickness, *t*, between points of substantial support, of plain furnaces or furnaces strengthened by stiffening rings, of flue sections and of the cylindrical bottoms of combustion chambers is to be determined by the following formulae the greater of the two thicknesses obtained being taken:

~~$t = \sqrt{\frac{pD_o(L+610)}{102\ 400}} + 0,75\ \text{mm}$~~ $t = \sqrt{\frac{pD_o(L+610)}{102\ 40}} + 0,75\ \text{mm}$

~~$t = \frac{C_pD_o}{1100} + \frac{L}{320} + 0,75\ \text{mm}$~~ $t = \frac{C_pD_o}{110} + \frac{L}{320} + 0,75\ \text{mm}$

11.3.2 Where stiffeners are used for strengthening plain cylindrical furnaces, or combustion chambers, the second moment of area, *I*, of the stiffener is to be determined by the following formula:

~~$I = \frac{pD^3oL}{13,3 \text{ x } 10^6}\text{mm}^4$~~ $I = \frac{pD_o3L}{13,3 \text{ x } 10^5}\ \text{mm}^4$

11.5 Hemispherical furnaces

11.5.1 The minimum thickness, *t*, of unsupported hemispherical furnaces subject to pressure on the convex surface is to be determined by the following formula:

~~$t = \frac{CpR_o}{608} + 0,75\ \text{mm}$~~ $t = \frac{CpR_o}{60,8} + 0,75\ \text{mm}$

11.6 Dished and flanged ends for supported vertical boiler furnaces

11.6.1 The minimum thickness, *t*, of dished and flanged ends for vertical boiler furnaces that are subject to pressure on the convex side and are supported by central uptakes, is to be determined by the following formula:

~~$t = \frac{pR_o}{10\sigma} + 0,75\ \text{mm}$~~ $t = \frac{pR_o}{\sigma} + 0,75\ \text{mm}$

11.7 Dished and flanged ends for unsupported vertical boiler furnaces

11.7.1 The minimum thickness, *t*, of dished and flanged ends for vertical boiler furnaces that are subject to pressure on the convex side and are without support from stays of any kind, is to be determined by the following formula, but is in no case to be less than the thickness of the firebox:

~~$t = \frac{CpR_o}{660} + 0,75\ \text{mm}$~~ $t = \frac{CpR_o}{66} + 0,75\ \text{mm}$

11.8 Ogee rings

11.8.1 The minimum thickness, *t*, of the ogee ring which connects the bottom of the furnace to the shell of a vertical boiler and sustains the whole vertical load on the furnace is to be determined by the following formula:

~~$t = \sqrt{\frac{pD_i(D_i - D_o)}{9900}} + 0,75\ \text{mm}$~~ $t = \sqrt{\frac{pD_i(D_i - D_o)}{990}} + 0,75\ \text{mm}$

11.9 Uptakes of vertical boilers

11.9.1 The minimum thickness, *t*, of internal uptakes of vertical boilers is to be determined by the following formulae, the greater of the two thicknesses obtained being taken:

~~$t = \sqrt{\frac{pD_o(L+610)}{102\ 400}} + 4\ \text{mm}$~~ $t = \sqrt{\frac{pD_o(L+610)}{102\ 40}} + 4\ \text{mm}$

~~$t = \frac{pD_o}{1100} + \frac{L}{320} + 4\ \text{mm}$~~ $t = \frac{pD_o}{110} + \frac{L}{320} + 4\ \text{mm}$

■ *Section 14*
Construction

14.1 Access arrangements

14.1.10 Circular flat cover plates may be fitted to raised circular manhole frames not exceeding 400 mm diameter, and for an approved design pressure not exceeding ~~18 bar~~ 1,8 MPa.

Part 5, Chapter 11
Other Pressure Vessels

■ *Section 2*
Cylindrical shells and drums subject to internal pressure

2.1 Minimum thickness

2.1.1 The minimum thickness, *t*, of a cylindrical shell is to be determined by the following formula:

~~$t = \frac{pR_i}{10\sigma J - 0{,}5p} + 0{,}75\text{ mm}$~~ $t = \frac{pR_{\text{i}}}{\sigma J - 0{,}5p} + 0{,}75\text{ mm}$

■ *Section 3*
Spherical shells subject to internal pressure

3.1 Minimum thickness

3.1.1 The minimum thickness, *t*, of a spherical shell is to be determined by the following formula:

~~$t = \frac{pR_i}{20\sigma J - 0{,}5p} + 0{,}75\text{mm}$~~ $t = \frac{pR_{\text{i}}}{2\sigma J - 0{,}5p} + 0{,}75\text{ mm}$

■ *Section 4*
Dished ends subject to internal pressure

4.1 Minimum thickness

4.1.1 The thickness, *t*, of semi-ellipsoidal and hemispherical unstayed ends, and the knuckle section of torispherical ends, dished from plate, having pressure on the concave side and satisfying the conditions listed below, is to be determined by the following formula:

~~$t = \frac{pD_oK}{20\sigma J} + 0{,}75\text{mm}$~~ $t = \frac{pD_{\text{o}}K}{2\sigma J} + 0{,}75\text{ mm}$

4.1.4 In addition to the formula in *Pt 5, Ch 11, 4.1 Minimum thickness 4.1.1* the thickness, *t*, of a torispherical head, made from more than one plate, in the crown section, is to be not less than that determined by the following formula:

~~$t = \frac{pR_i}{20\sigma J - 0{,}5p} + 0{,}75\text{mm}$~~ $t = \frac{pR_i}{2\sigma J - 0{,}5p} + 0{,}75\text{ mm}$

■ *Section 5*
Dished ends for Class 3 pressure vessels

5.1 Minimum thickness

5.1.1 As an alternative to the formula in *Pt 5, Ch 11, 4.1 Minimum thickness 4.1.1*, for Class 3 vessels only, the minimum thickness, *t*, of a torispherical unstayed end dished from plate and having pressure on the concave or convex side is to be determined by the following formula:

~~$t = \frac{pR_i}{CS}$~~ $t = \frac{10pR_{\text{i}}}{CS}$

■ *Section 6*
Conical ends subject to internal pressure

6.2 Minimum thickness

6.2.1 The minimum thickness, *t*, of the cylinder, knuckle and conical section at the junction and within the distance *L* from the junction is to be determined by the following formula:

~~$t = \frac{pD_oK}{20\sigma J} + 0{,}75\text{mm}$~~ $t = \frac{pD_oK}{2\sigma J} + 0{,}75\text{ mm}$

6.2.3 The minimum thickness, *t*, of those parts of conical sections not less than a distance *L* from the junction with a cylinder or other conical section, is to be determined by the following formula:

~~$t = \frac{pD_c}{20\sigma J - p}\frac{1}{\cos\alpha} + 0{,}75\text{ mm}$~~ $t = \frac{pD_c}{2\sigma J - p}\frac{1}{\cos\alpha} + 0{,}75\text{ mm}$

■ *Section 7*
Standpipes and branches

7.1 Minimum thickness

7.1.1 The minimum wall thickness, *t*, of standpipes and branches is to be not less than the greater of the two values determined by the following formulae, making such additions as may be necessary on account of bending, static loads and vibrations:

~~$t = \frac{pD_o}{20\sigma + p} + 0{,}75\text{ mm, or}$~~ $t = \frac{pD_o}{2\sigma + p} + 0{,}75\text{ mm, or}$

■ *Section 10*
Hydraulic tests

10.1 General

10.1.1 Pressure vessels covered by this Chapter are to be tested on completion to a pressure, $p_{-\mathrm{T}}$, determined by the following formula, without showing signs of weakness or defect:

$$P_{\mathrm{T}} = 0{,}13\frac{\sigma_{50}t}{\sigma_{\mathrm{T}}(t-0{,}75)}p$$

$\neq$ but in no case is to exceed

$$= 0{,}15\frac{t}{(t-0{,}75)}p$$

where

p = design pressure, in ~~bar~~ MPa

p_{T} = test pressure, in ~~bar~~ MPa

■ *Section 11*
Plate heat exchangers

11.1 General

11.1.1 Plate heat exchangers are to be classed as follows. Class 2 where either of the following conditions apply:

(b) design pressure is ~~17,2 bar~~ 1,72 MPa or greater.

Part 5, Chapter 12
Piping Design Requirements

■ *Section 2*
Carbon and low alloy steels

2.1 Carbon and low alloy steel pipes, valves and fittings

2.1.1 Materials for Class I and Class II piping systems, also for ship-side valves and fittings and valves on the collision bulkhead, are to be manufactured and tested in accordance with the appropriate requirements of ~~the Rules for Materials, *see also*~~ *Pt 5, Ch 12, 1.7 Materials* of the *Rules for the Manufacture, Testing and Certification of Materials, July 2019*.

2.1.2 Materials for Class III piping systems are to be manufactured and tested in accordance with the requirements of acceptable national specifications. Pipes having forge butt welded longitudinal seams are not to be used for fuel oil systems, for heating coils in oil tanks, or for pressures exceeding ~~4,0 bar (4,1 kgf/cm^2~~) 0,4 MPa. The manufacturer's certificate will be acceptable and is to be provided for each consignment of material. *See Ch 1, 3.1 General 3.1.3.(c)* of the ~~Rules for Materials~~ *Rules for the Manufacture, Testing and Certification of Materials, July 2019*.

2.1.3 Steel pipes, valves and fittings may be used within the temperature limits indicated in T*able 12.2.1 Carbon and carbon-manganese steel pipes* and *Table 12.2.2 Alloy steel pipes*. Where rimming steel is used for pipes manufactured by electric resistance or induction welding processes, the design temperature is limited to 400°C, *see Ch 6, 3 Welded pressure pipes* of the ~~Rules for Materials~~ *Rules for the Manufacture, Testing and Certification of Materials, July 2019*.

2.5 Screwed-on flanges

2.5.3 Such screwed and expanded flanges may be used for steam for a maximum design pressure of ~~30,0 bar (30,5 kgf/cm^2)~~ 3 MPa and a maximum design temperature of 370°C and for feed for a maximum design pressure of ~~50 bar (51 kgf/cm^2).~~ 5 MPa.

2.12 Other mechanical couplings

2.12.4 Mechanical joints for pressure pipes are to be tested to a burst pressure of 4 times the design pressure. For design pressures above ~~200 bar~~ 20 MPa the required burst pressure will be specially considered.

2.12.10 Restrained slip-on joints are permitted in steam pipes with a design pressure of ~~10 bar~~ 1 MPa or less on the weather decks of oil and chemical tankers to accommodate axial pipe movement, *see Pt 5, Ch 13, 2.7 Provision for expansion*.

2.14 Carbon dioxide (CO_2) fire-extinguishing system piping

2.14.2 Materials for the distribution manifolds between the carbon dioxide storage bottles and the discharge valves to each section and associated pipes, valves and fittings of high pressure systems are to be manufactured and tested in accordance with the requirements for Class I piping systems. Pipes are to meet the minimum wall thickness requirements of *Table 12.2.9 Minimum thickness for steel pipes for CO2 fire-extinguishing* and the manifold system is to be hydraulically tested to a pressure of ~~190 bar~~ 19 MPa. A high pressure system is defined as a system where the carbon dioxide is stored at ambient temperature.
Materials for the distribution manifolds between the carbon dioxide storage vessel(s) and the discharge valves to each section and associated pipes, valves and fittings of low pressure systems are to be manufactured and tested in accordance with the requirements for Class II piping systems and the manifold system is to be hydraulically tested to a pressure of ~~33 bar~~ 3,3 MPa. A low pressure system is defined as a system where the carbon dioxide is stored at a working pressure in the range of ~~18 bar to 22 bar~~ 1,8 to 2,2 MPa.

2.14.3 Piping downstream of the distribution valve(s) for high pressure systems is to be manufactured and tested in accordance with the requirements for Class II piping and is to meet the minimum wall thickness requirements of *Table 12.2.9 Minimum thickness for steel pipes for CO2 fire-extinguishing*. After installation the distribution system is to be leak tested at a pressure of ~~6 bar~~ 0,6 MPa. Piping downstream of the distribution valve(s) for low pressure systems is to be manufactured and tested in accordance with the requirements for Class III piping. After installation the distribution system is to be leak tested at a pressure of ~~6 bar~~ 0,6 MPa. Class III piping may be used for open ended distribution piping downstream of the distribution valve(s) of high pressure systems where agreed by LR and where meeting the minimum wall thickness requirements of *Table 12.2.9 Minimum thickness for steel pipes for CO2 fire-extinguishing* and where a minimum of ten per cent of the piping is hydraulically tested at a pressure of ~~125 bar~~ 12, 5 MPa. This testing is to be carried out before installation.

2.14.5 If it is necessary for carbon dioxide pipes to pass through accommodation spaces, the pipe is to be seamless and is to meet the requirements for Class II pipes. Joints are to be made only by welding and the pipes are to be hydraulically tested after installation at a pressure of ~~-50 bar~~ 5 MPa.

■ *Section 4*
Cast iron

4.2 Grey cast iron

4.2.3 Grey cast iron is not to be used for the following:

- Piping system components in flammable oil systems where the design pressure exceeds ~~7 bar~~ 0,7 MPa or the design operating temperature is greater than 60°C.

■ *Section 5*
Plastic Pipes

5.5 Additional fire performance criteria applicable to ships

Table 12.5.1 Fire endurance requirements

Location											
	A	B	C	D	E	F	G	H	I	J	K
MISCELLANEOUS											
31 Auxiliary low pressure steam (≤ ~~7 bar~~ 0,7 MPa)	L2	L2	0[9]	0[9]	0[9]	0	0	0	0	0[9]	0[9]

■ *Section 7*
Flexible hoses

7.1 General

7.1.3 The use of hose clamps and similar types of end attachments are not to be used for flexible hoses in piping systems for steam, flammable media, starting air systems or for sea-water systems where failure may result in flooding. In other piping systems, the use of hose clamps may be accepted where the working pressure is less than ~~5 bar~~ 0,5 MPa and provided that there are two clamps at each end connection.

■ *Section 8*
Hydraulic tests on pipes and fittings

8.1 Hydraulic tests before installation on board

8.1.1 All Class I and II pipes and their associated fittings are to be tested by hydraulic pressure to the Surveyor’s satisfaction. Further, all steam, feed, compressed air and fuel oil pipes, together with their fittings, are to be similarly tested where the design pressure is greater than ~~7bar~~ 0,7 MPa. The test is to be carried out after completion of manufacture and before installation on board and, where applicable, before insulating and coating.

8.1.4 For steel pipes and integral fittings for use in systems where the design temperature exceeds 300°C, the test pressure is to be as follows:

(b) For alloy steel pipes, the test pressure is to be determined by the following formula, but need not exceed 2*p*:

$Pt = 1{,}5 \frac{\sigma_{100}}{\sigma} p$ ~~bar(kgf/cm^2)~~ MPa

= where p_t, and *p* are as defined in *Pt 5, Ch 12, 1.3 Design symbols 1.3.2*

σ = permissible stress for the design temperature, in N/mm^2 (~~kgf/cm^2)~~, as stated in *Table 12.2.2 Alloy steel pipes*.

σ_{100} = permissible stress for 100°C, in N/mm^2 ~~(kgf/cm^2),~~ as stated in *Table 12.2.2 Alloy steel pipes*.

8.2 Testing after assembly on board

8.2.1 Heating coils in tanks, gas fuel and fuel oil piping are to be tested by hydraulic pressure, after installation on board, to 1,5 times the design pressure but in no case to less than ~~4 bar (4,1 kgf/cm^2)~~ 0,4 MPa.

Part 5, Chapter 13
Ship Piping Systems

■ *Section 1*
General requirements

1.3 Plans and particulars

1.3.1 The following plans (in diagrammatic form) and particulars are to be submitted for approval. Additional plans should not be submitted unless the arrangements are of a novel or special character affecting classification:

(v) Arrangements and dimensions of all steam pipes where the design pressure or temperature exceeds ~~16,0 bar (16,3 kgf/cm^2)~~ 1,6 MPa or 300°C, respectively, and the outside diameter exceeds 76,1 mm, with details of flanges, bolts and weld attachments, and particulars of the material of pipes, flanges, bolts and electrodes.

■ *Section 2*

Construction and installation

2.5 Ship-side valves and fittings (other than those on scuppers and sanitary discharges)

2.5.10 Valves, cocks and distance pieces, intended for installation on the ship's side below the load waterline, are to be tested by hydraulic pressure to not less than ~~5 bar~~ 0,5 MPa.

Part 5, Chapter 14
Machinery Piping Systems

■ *Section 4*

Fuel oil pumps, pipes, fittings, tanks, etc.

4.5 Pipes conveying oil

4.5.2 Where pipes convey heated oil under pressure the flanges are to be machined, and the jointing material, which is to be impervious to oil heated to 150°C, is to be the thinnest possible, so that flanges are practically metal to metal. The scantlings of the pipes and their flanges are to be suitable for a pressure of at least ~~13,7 bar (14 kgf/cm^2)~~ 1,37 MPa or for the design pressure, whichever is the greater.

4.6 Low pressure pipes

4.6.1 Transfer, suction and other low pressure oil pipes and all pipes passing through oil storage tanks are to be made of cast iron or steel, having flanged joints suitable for a working pressure of not less than ~~6,9 bar (7 kgf/cm^2)~~ 0,69 MPa. The flanges are to be machined and the jointing material is to be impervious to oil. Where the pipes are 25 mm bore or less, they may be of seamless copper or copper alloy, except those which pass through oil storage tanks. Oil pipes within the engine and boiler spaces are to be fitted where they can be readily inspected and repaired.

4.10 Relief valves on oil heaters

4.10.1 Relief valves are to be fitted on the oil side of heaters and are to be adjusted to operate at a pressure of ~~3,4 bar (3,5 kgf/cm^2)~~ 0,34 MPa above that of the supply pump relief valve, *see Pt 5, Ch 14, 4.3 Relief valves on pumps*. The discharge from the relief valves is to be led to a safe position.

■ *Section 10*

Low pressure compressed air systems

10.1 General

10.1.2 Low pressure compressed air systems are to produce and distribute cooled compressed air throughout the ship to supply all pneumatic control and instrumentation systems where the air pressure requirements are typically ~~3 to 10 bar~~ 0,3 to 1 MPa. LP compressed air systems may include air compressors, oil/water separators, filters, dryers, distribution lines and air receivers.

Part 5, Chapter 15
Piping Systems for Oil Tankers

■ *Section 3*

Cargo handling system

3.2 Cargo pumps

3.2.4 Where cargo pumps are driven by shafting which passes through a pump-room bulkhead or deck, gastight glands are to be fitted to the shaft at the pump-room plating. The glands are to be efficiently lubricated from outside the pump-room. The seal parts of the glands are to be of materials that will not initiate sparks. The glands are to be of an approved type and are to be attached to the bulkhead in accordance with *Pt 5, Ch 13, 2.4 Attachment of valves to watertight plating*. Where a bellows piece is incorporated in the design, it is to be hydraulically tested to ~~3,4 bar (3,5 kgf/cm^2)~~ 0,34 MPa before fitting.

■ *Section 4*

Cargo tank venting, purging and gas-freeing

4.1 Cargo tank venting

4.1.14 Pressure/vacuum valves are to be set at a positive pressure of not more than ~~0,2 bar (0,2 kgf/cm^2)~~ 0,02 MPa above atmospheric and a negative pressure of not more than ~~0,07 bar (0,07 kgf/cm^2)~~ 0,007 MPa below atmospheric. Higher positive pressures not exceeding ~~0,7 bar (0,7 kgf/cm^2)~~ 0,07 MPa gauge may be permitted in specially designed integral tanks.

Part 5, Chapter 16

Water ~~jet~~ Jet Systems

■ *Section 7*

Inspection, testing and fitting of water jets

7.2 Shop tests and installation of water jet systems

7.2.1 The completed water jet unit is to undergo a tightness test in which an internal hydrostatic pressure of ~~1,5 bar~~ 0,15 MPa above the maximum working pressure of the unit is to be applied.

Part 6, Chapter 3

Refrigerated Cargo Installations

■ *Section 1*

General requirements

1.4 Equipment to be constructed under survey

1.4.1 All major items of equipment are to be surveyed at the manufacturer's works. The workmanship is to be to the Surveyor's satisfaction and the Surveyor is to be satisfied that the components are suitable for the intended purpose and duty. Examples of such units are:

- Valves and other components intended for installation in pressure piping systems having a maximum working pressure greater than ~~7 bar~~ 0,7 MPa.

■ *Section 4*

Refrigeration plant, pipes, valves and fittings

4.15 Overpressure protection devices

4.15.7 Each compressor is to be provided with automatic shutdown in the event of high discharge pressure. For refrigeration systems where the maximum working pressure is less than or equal to ~~40 bar g~~ 4 MPa, the automatic shutdown is to operate at a pressure in excess of normal operating pressure but no greater than 0,9 of the maximum working pressure. For refrigeration systems where the maximum working pressure is greater than ~~40 bar g~~ 4 MPa, the automatic shutdown is to operate at a pressure in excess of normal operating pressure but no greater than 0,95 of the maximum working pressure.

Part 7, Chapter 10
Carriage of Refrigerated Containers

■ *Section 6*
Hold access and maintenance access arrangements

6.1 Hold pressure/vacuum

6.1.1 The maximum permitted pressure or vacuum that may occur in the hold space is to be stated. It is proposed that a value, in accordance with the contents of *Pt 3, Ch 1, 9.7 Definitions and details of tests 9.7.4*, may be considered as a maximum value. An overpressure of ~~0,15 bar~~ 0,015 MPa may be used for guidance purposes. If the ventilation system is capable of producing a positive pressure or vacuum in excess of the design allowable figure, then means are to be provided to protect the hold space against the effect of over pressure or vacuum. If axial supply fans are proposed, even if aerofoil fan blades are fitted, it is unlikely that the fans will be able to produce a pressure above ~~0,025 bar~~ 0,0025 MPa (250 mm water gauge).

■ *Section 9*
Inspection and testing on completion

9.3 Testing of cooling water system

9.3.1 Cooling water piping that is welded *in situ* is to be hydraulically strength tested at 1,5 times the design pressure, but in no case less than ~~3,5 bar g~~ 0,35 MPa.

Part 7, Chapter 15
Refrigeration Systems and Equipment Serving Provision Stores and Air-Conditioning Installations

■ *Section 2*
Construction and installation

2.7 Design pressures

Table 15.2.1 Design pressure limits

Refrigerant	Pressure, (~~bar g~~) (MPa)	
	High	Low
R-717	~~21,2~~ 2,12	~~17,2~~ 1,72
R-22	~~20,6~~ 2,06	~~16,7~~ 1,67
R-290	~~18,1~~ 1,81	~~14,7~~ 1,47
R-600a	~~6,4~~ 0,64	~~5,2~~ 0,52
R134a	~~13,4~~ 1,34	~~10,9~~ 1,09
R-407C	~~23,5~~ 2,35	~~19,0~~ 1,9
R-410A	~~34,5~~ 3,45	~~28,0~~ 2,8
R-507A	~~25,3~~ 2,53	~~20,5~~ 2,05
R-404A	~~24,8~~ 2,48	~~20,1~~ 2,01

■ *Section 3*
Refrigerating machinery and refrigerant storage compartment arrangements

3.3 Pressure testing at manufacturer's work

Table 15.3.1 Test pressure

Component	Test pressure, (~~bar g~~) (MPa)	Tightness test
	Pressure test	
(1) Pressure vessels	*See Pt 5, Ch 11, Other Pressure Vessels*	1,0*p*
(2) Compressor cylinders/crankcase/casing	1,5p	1,0*p*
(3) Valves and fittings	2,0p	1,0*p*
(4) Pressure piping, fabricated~~.~~, headers, air coolers, etc	1,5p	1,0*p*

3.3.4 Components for use with a secondary refrigerant or cooling water are to be hydraulically tested to 1,5 times the design pressure which, in no case is to be less than ~~3,5 bar g~~ 0,35 MPa.

3.4 Pressure test after installation on board ship

3.4.5 Secondary refrigerant piping welded in place is to be hydraulically tested to 1,5 times the design pressure which, in no case is to be less than ~~3,5 bar g~~ 0,35 MPa.

■ *Section 5*
Control and monitoring and electrical power arrangements

5.1 General

5.1.13 Compressors are to be provided with automatic shutdown in the event of high discharge pressure. For refrigeration systems where the maximum working pressure is less than or equal to ~~40 bar g~~ 4 MPa~~.~~, the automatic shutdown is to operate at a pressure in excess of normal operating pressure but no greater than 0,9 of the maximum working pressure. For refrigeration systems where the maximum working pressure is greater than ~~40 bar g~~ 4 MPa, the automatic shutdown is to operate at a pressure in excess of normal operating pressure but no greater than 0,95 of the maximum working pressure.

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